



Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Thermal analysis of a compressor for application to Compressed Air Energy Storage

C. Zhang^a, B. Yan^a, J. Wieberdink^a, P.Y. Li^a, J.D. Van de Ven^a, E. Loth^b, T.W. Simon^{a,*}

^a Mechanical Engineering Department, University of Minnesota, 111 Church St. S.E. Minneapolis, MN 55455, USA

^b Mechanical and Aerospace Engineering Department, University of Virginia, 122 Engineer's Way P.O. Box 400746, Charlottesville, VA 22904, USA

HIGHLIGHTS

- We analyze heat transfer enhancement in a liquid-piston compressor for CAES.
- Hydrothermal characterizations of the inserted heat exchanger media are obtained.
- CFD simulations are done combining the VOF and porous media modeling methods.
- The heat exchangers can effectively suppress temperature rise and secondary flows.
- Results indicate optimization on the shape of the heat exchanger could be done.

ARTICLE INFO

Article history:

Received 16 January 2014

Received in revised form

24 July 2014

Accepted 3 August 2014

Available online xxx

Keywords:

Energy storage

Compressor

Wind turbine

Heat transfer

Porous media

ABSTRACT

In this paper, the topic of Compressed Air Energy Storage (CAES) is discussed and a program in which it is being applied to a wind turbine system for leveling power supplied to the grid is described. Noted is the importance of heat transfer in the design of the compressor and its effect on performance. Presented is a design for minimizing the temperature rise in the compressor during compression. The design requires modeling regenerative heat transfer from the compressed air to solid material inserted in the compression space. Modeling requires characterizing pressure drop through the porous insert, interfacial heat transfer between solid and fluid in the matrix, and thermal dispersion within the porous regions. Computation and experimentation are applied for developing correlations for such terms. Two types of porous media are applied: interrupted plates and open-cell metal foams. Cases with foam inserts are computed and the results are discussed. Discovered in the results are some complex secondary flow features in spaces above the porous inserts.

© 2014 Elsevier Ltd. All rights reserved.

1. Introduction

This paper presents thermal analyses on a liquid piston driven compressor used for Compressed Air Energy Storage (CAES). The CAES system stores energy as high-pressure air, to retrieve it later in a liquid piston expander. Compression leads to a tendency for temperature rise in a compressible gas. Absorbing heat from air during compression to reduce its temperature rise is important for improving compression efficiency. As the air temperature rises, part of the input work is being converted into internal energy rise that is wasted during the storage period as the compressed air cools toward the ambient temperature. This paper discusses techniques for minimizing that temperature rise.

Our desire is that the CAES system is integrated into a wind power field. The benefit of integrating energy storage into a wind power array is that fluctuations in wind power input can be smoothed over time and electric power generation equipment can be sized commensurate with a supply power that is nearer the average wind power of the day. The alternative is sizing for peak power or “throttling back” the wind turbine when the wind is strong.

Use of storage techniques for wind power is discussed in Ref. [1]. Various scenarios with and without storage were considered. In one, wind generation capacity in 2050 was 302 GW without storage and 351 GW with storage. Another study, which is modeled using CAES and wind energy to supply 20% of the total power generation in Texas concluded that the addition of CAES to wind energy reduces the impact of wind fluctuations on the electricity grid [2]. From a report written by the US Offshore Wind Collaborative [3], we read that the total amount of U.S. offshore wind energy capacity

* Corresponding author.

E-mail address: tsimon@me.umn.edu (T.W. Simon).

Nomenclature

a_v	surface area per unit volume of porous medium (1/m)
A	cross-sectional area (m ²)
b	half plate distance (m)
$\mathcal{C}$	coefficient for the Forchheimer term (1/m)
c_p	constant-pressure specific heat (J/kg K)
c_s	specific heat for solid (J/kg K)
c_v	constant-volume specific heat (J/kg K)
D_h	hydraulic diameter (m)
D	characteristic diameter (m)
d	characteristic length based on filament dia. (m)
d_f	filament diameter (m)
d_m	mean pore diameter (m)
E_s	storage energy (J)
$\vec{g}$	gravitational acceleration (m/s ²)
h_{sf}	surface heat transfer coefficient (W/m ² K)
h_v	volumetric heat transfer coefficient (W/m ³ K)
K	permeability (m ²)
k	thermal conductivity (W/m K)
k_{dis}	dispersion conductivity (W/m K)
L	chamber length (m)
L_1	length of the upper region without insert (m)
L_{ins}	length of the insert region (m)
l	plate length (m)
m	total mass of air (kg)
n	polytropic exponent
Nu	Nusselt number
P	bulk pressure (Pa)
P_{tot}	supplied hydraulic pressure (Pa)
Pe	Peclet number
$\bar{q}$	area averaged wall heat flux
p	local pressure (Pa)
P_f	final air pressure (Pa)
Pr	Prandtl number
R	radius of chamber (m)
Re_l	Reynolds number based on l
r	radial coordinate
$\mathcal{R}$	ideal gas constant (J/kg K)

Re	Reynolds number based on characteristic length d
$\vec{S}_m$	momentum source term (Pa/m)
T	local air temperature (K)
T_0	initial temperature; wall temperature (K)
T_a	average air temperature in the chamber (K)
T_s	local solid temperature (K)
T_{solid}	average temperature of solid in the chamber (K)
T_f	final averaged air temperature (K)
t	plate thickness (m)
τ	time (s)
U_{in}	liquid piston velocity (m/s)
V	instantaneous volume of chamber (m ³)
W_{in}	work input (J)
x	axial coordinate

Greek symbols

α	volume fraction
ε	porosity
η	compression efficiency
μ	dynamic viscosity (Ns/m ²)
ρ	density (kg/m ³)
ζ	final pressure compression ratio

Subscripts

0	initial value of variable
1	air phase
2	water phase
D	based on filament diameter
Da	Darcian
f	fluid phase
REV	averaged on the REV
s	solid
XX	local streamwise direction
YY	local cross-stream direction

Superscripts

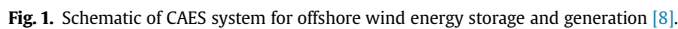
*	dimensionless variable
---	------------------------

is almost equal to the current total installed capacity. Noting that most electricity-demanding regions are on the coasts, the report highlights the significant potential for offshore wind generation. Traditional CAES systems store the compressed air in underground caverns and often marry this with combustion plants by using compressed air as oxidant for burning in combustion turbines [4]. More recent advancements include using multi-stage compressors with inter-cooling between stages and multi-stage expansion turbines, with reheat [5]. A small-scale wind turbine system with CAES was constructed and tested [6]. Although the study showed beneficial prospects of using combined CAES with the wind turbine system for home applications, the efficiency of the system tested was very low, mainly due to a lack of cooling during compression. Herein, near-isothermal compression and expansion are by using regenerative heat exchange with elements *within* the compressor/expander volumes.

In the present system, we employ the “open accumulator” concept proposed in Ref. [7]. In this system, gas is exhausted to the atmosphere during expansion, reversing during compression. Compression is with pumped liquid and air and expansion is with a liquid motor and an air expander. It is operated at a constant

accumulator pressure; the maximum design pressure. Thus, with the open accumulator, the energy storage density per unit volume of air is always high since the low-power, low-accumulator-pressure situation in the closed accumulator is avoided. An added benefit is a dramatic reduction in accumulator pressure oscillation cycles and improved fatigue performance.

Though not necessary for the open accumulator CAES to be successful, our design is proposed to be used in a fluid power wind turbine system. In such a system, we use hydraulic equipment for power conversion and transmission, as shown in Fig. 1. It employs hydraulic circuits, a hydraulic drive pump, hydraulic pumps and motors, open accumulators, and liquid-piston air compressors [8]. Advantages include having a hydraulic pump in the nacelle at the top of the tower rather than a heavy transmission and generator assembly and having much of the equipment, including the generator, at ground (or sea) level, allowing easier access for assembly and maintenance. When electricity demand is small, the system operates in storage mode in which the excess shaft power from the wind turbine is transferred to hydraulic power in the drive pump and the hydraulic power is transmitted through the hydraulic circuit to the hydraulic transformers, which are a series of



Key to success of the system is efficient compression and expansion of air in the compressor/expander. As noted, this must be done under near-isothermal conditions. Thus, the present paper will concentrate on the thermal design of the compressor/expander and the analysis on which it is based. This paper focuses on compression and expansion with a liquid piston. The term “liquid piston” is applied because the gas compression is done with a liquid–gas interface rather than a solid piston. It is found in Ref. [9] that liquid-piston compressors have an advantage over solid pistons in terms of efficiency and power consumption. Further, for the purpose of cooling, insertion of solid material into the chamber is possible with liquid-piston compressors as liquid can flow through the open portions of the matrix. The use of porous media and regenerative heat exchangers to absorb thermal energy has been shown to be successful in a number of applications, such as a micro-fabricated, segmented, involute-foil regenerator used in a Stirling engine [10], a honeycomb regenerator used in a high-temperature-air-combustion system [11], a regenerative heat exchanger used in a reverse-flow reactor [12], and a magnetic regenerative system used in a refrigeration cycle [13]. In the present study, it will be shown that having a liquid piston design for the compressor inserted with porous media is a major step in having an efficient compressor/expander. The remainder of the paper will be dedicated to modeling thermal storage to the porous material in the compressor volume in preparation for design of the compressor.

The effect of heat transfer on compression efficiency will be shown quantitatively through a simple thermodynamic analysis.

Fig. 2. Effects of heat transfer on compression efficiency with different pressure compression ratios.

3. Heat transfer to the porous inserts

3.1. Introduction to porous inserts

Porous media can be inserted into the liquid-piston compression chamber so that the solid can absorb thermal energy from the air as it heats during compression. The present study will introduce two kinds of porous inserts, an open-cell metal foam and an interrupted-plate insert (Fig. 3). The metal foam has very thin filament features that enhance mixing of the flow. The interrupted-plate insert consists of an array of plates oriented in a staggered fashion so that a new thermal boundary layer develops on each successive plate in the streamwise direction. Both have large heat transfer surface area to volume ratios. The open-cell metal foam is a common type of regenerative heat exchanger. The interrupted-plate insert provides opportunities for us to design its shape to result in improvements; the form of the matrix is chosen, but then different sizes of the shape parameters are studied to adjust the shape.

The flow resistance and resulting pressure drop, the heat transfer coefficient between the solid material and surrounding air, and thermal dispersion will be discussed for the two types of porous inserts discussed in this paper. Then, using these models as source terms in the transport equations, simulations on the full compressor with a metal foam insert will be done.

3.2. Interrupted plate

Because details of the shape of the interrupted-plate insert are known, CFD simulations on a Representative Elementary Volume

(REV) of the interrupted-plate can be done. These CFD analyses are used to obtain the pressure drop and interfacial heat transfer relationships for flow with various velocities through the matrix. The REV is the minimum representative, repeating geometric feature of the porous medium. Its geometry for the interrupted plate insert is shown in Fig. 3(c). Three geometric dimensions are: the plate length, l , the plate distance, $2b$, and the plate thickness, t .

3.2.1. Shape analysis

A study was conducted to quantify the effects of the shape parameters on heat transfer and compression efficiency [16]. Twenty-seven different shapes were analyzed. For each shape, two flow conditions were studied: $Re_1 = 1$ and $Re_1 = 400$, where Re_1 is defined as.

$$Re_1 = \frac{\rho u_{REV,f}}{\mu_{REV,f}} \quad (6)$$

The flow is presumed to be in the same direction as the x axis (Fig. 3(c)). Periodic momentum and thermal boundary conditions are used on the entry and exit surfaces of the REV model such that the flow being simulated is representative of a fully-developed flow through the central region of the porous matrix. The solid surfaces are maintained at a uniform and constant temperature in the REV simulations.

From these CFD simulation results, a heat transfer correlation, generalized in terms of hydraulic diameter, is developed. The Reynolds number and Nusselt number based on hydraulic diameter are:

$$Re_{Dh} = \frac{\rho D_h u_{REV,f}}{\mu_{REV,f}} \quad (7)$$

$$Nu_{Dh} = \frac{h D_h}{k} \quad (8)$$

where,

$$h = \frac{\bar{q}}{T_0 - T_{bulk,f}} \quad (9)$$

$$Dh = 4b \quad (10)$$

The heat transfer correlation is given by Ref. [16]:

$$Nu_{Dh} = 9.700 + 0.0876 Re_{Dh}^{0.792} Pr^{\frac{1}{3}} \quad (11)$$

It is plotted with the data from the various CFD runs in Fig. 4(a). As the shape parameters in the CFD runs can be varied, this correlation captures the effects of different shapes on heat transfer.

The pressure drop for flow passing through the interrupted-plate exchanger can be modeled using a viscous and an inertial term.

$$\frac{\partial p}{\partial x} = -\frac{\mu}{K} u_{REV} - \frac{F}{K^{0.5}} \rho u_{REV}^2 \quad (12)$$

Using data from the study given in Refs. [16], the permeability and Forchheimer coefficient for a given plate geometry is correlated against the plate length and plate thickness by a least square fit:

$$K_{2.5} = \left[0.1151 \lg\left(\frac{l}{1\text{m}}\right) - 5.184 \frac{l}{1\text{m}} - 173.9 \frac{t}{1\text{m}} + 1.031 \right] \times 10^{-6} \text{ m}^2 \quad (13)$$

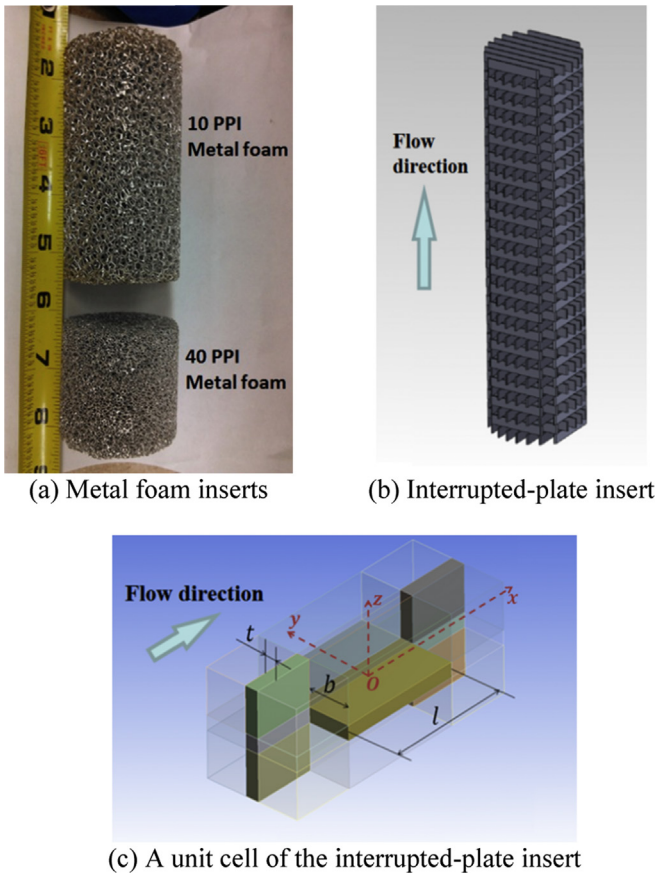


Fig. 3. Porous inserts used in the liquid piston compressor.

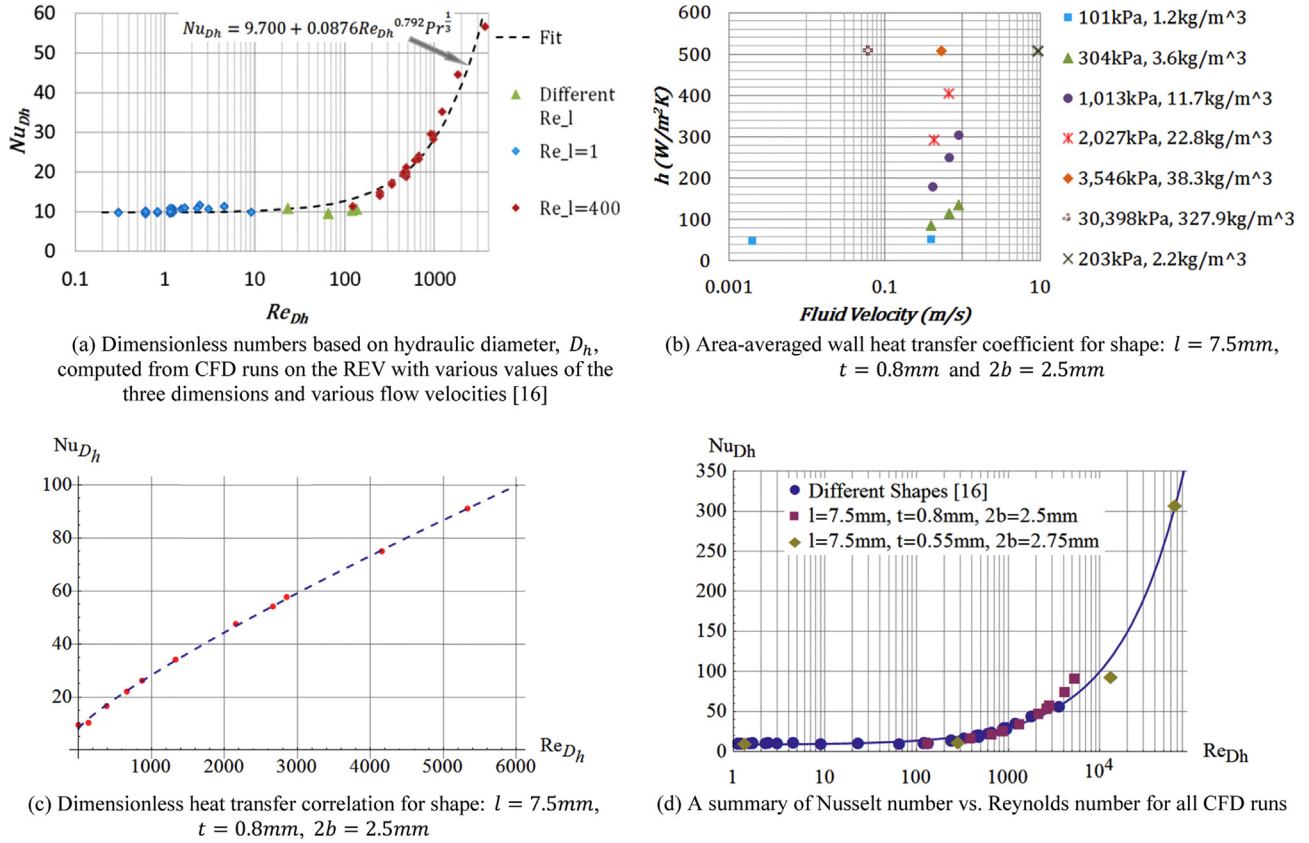


Fig. 4. Various heat transfer simulation studies on the REV of the interrupted-plate exchanger.

$$F_{2.5} = -0.009979 \lg\left(\frac{l}{1\text{m}}\right) + 62.64\left(\frac{t}{1\text{m}}\right)^{1.057} - 0.04957 \quad (14)$$

$$K_5 = \left[0.3449 \lg\left(\frac{l}{1\text{m}}\right) + 1.285 \frac{l}{1\text{m}} - 394.0 \frac{t}{1\text{m}} + 3.195 \right] \times 10^{-6} \text{m}^2 \quad (15)$$

$$F_5 = 0.003209 \lg\left(\frac{l}{1\text{m}}\right) - 1.134 \frac{l}{1\text{m}} + 403.0\left(\frac{t}{1\text{m}}\right)^{1.493} + 0.03676 \quad (16)$$

$$K_{10} = \left[-0.03438 \lg\left(\frac{l}{1\text{m}}\right) + 155.3 \frac{l}{1\text{m}} - 984.2 \frac{t}{1\text{m}} + 3.835 \right] \times 10^{-6} \text{m}^2 \quad (17)$$

$$F_{10} = 0.009688 \lg\left(\frac{l}{1\text{m}}\right) - 1.319 \frac{l}{1\text{m}} + 9.397 \frac{t}{1\text{m}} + 0.06492 \quad (18)$$

where subscripts 2.5, 5 and 10 represent plate lengths of 2.5 mm, 5 mm and 10 mm, respectively.

Using the interfacial heat transfer correlation, the compression process is simulated based on a zero-dimensional, thermodynamic model, from which the efficiency (Eq. (3)) is evaluated for each shape [16]. It is found that, in general, decreasing the separation

distance between plates in the interrupted plate insert leads to improved efficiency. For a given plate separation distance, decreasing the plate length improves efficiency, as shown from a simple boundary layer development analysis. The heat transfer correlation and the pressure drop equations developed from this study can be used also in optimization of the exchanger shape.

3.2.2. Heat transfer and pressure drop under high Reynolds number

Analyses in this section are focused on heat transfer under relatively high Reynolds number flow. Simulations are done at eleven different Reynolds numbers, Re_{Dh} , ranging from 0.67 to 5333, for an interrupted-plate insert with the following shape parameters: $l = 7.5\text{mm}$, $t = 0.8\text{mm}$, $2b = 2.5\text{mm}$. Three different runs are computed at $Re_{Dh} = 5333$ while varying the density of the air flow and the mass flow rate. Each is a single CFD run. The commercial CFD Software ANSYS FLUENT is used and the $k-\epsilon$ model is used for turbulence closure. The REV models have periodic momentum and thermal boundary conditions. Details of implementing these conditions are discussed in Ref. [16]. The number of computational cells varies with Reynolds number; specifically: 169,632 for $Re_{Dh} < 150$, 588,816 for $150 \leq Re_{Dh} < 2000$, 1,357,056 for $2000 \leq Re_{Dh} < 5000$, and 2,024,352 for $Re_{Dh} \geq 5000$. These grid cells have been verified by grid-independence studies. From the CFD simulations, heat transfer coefficients and pressure drop terms are obtained.

The area-averaged surface heat transfer coefficient depends on both flow velocity and density, as shown in Fig. 4(b). The Reynolds numbers and Nusselt numbers can very well capture heat transfer under different velocity and pressure conditions, as shown in Fig. 4(c). The permeability and Forchheimer coefficient for this shape are also computed:

$$K = 2.813 \times 10^{-7} \text{ m}^2, \quad F = 0.0359$$

Another set of simulations are done at four Reynolds numbers for an interrupted-plate insert with shape parameters: $l = 7.5 \text{ mm}$, $t = 0.55 \text{ mm}$, $2b = 2.75 \text{ mm}$. For $Re_{Dh} = 1.3$ and 286.9 , a grid of $707,840$ cells is used and the laminar flow is solved. For $Re_{Dh} = 13,415.6$ and $66,820.6$, the grid cells are, respectively, $1,469,139$ and $2,584,833$, and the Transition SST model (see Ref. [17]) is used for turbulence modeling. The Nusselt numbers and Reynolds numbers of these simulations are compared with results from all previous simulations on the basis of hydraulic diameter in Fig. 4(d). It can be seen that at small Reynolds numbers (roughly $Re_{Dh} \leq 5000$), all data represented tend to collapse on a hydraulic-diameter-based correlation even though other shape parameters are different. Yet, this tendency is weakened as Re_{Dh} becomes larger. A least square fit of all data points representing Reynolds numbers ranging from 1 to $66,820.6$ yields the following correlation, which is shown by the curve in Fig. 4(d).

$$Nu_{Dh} = 8.456 + 0.325 Re_{Dh}^{0.625} Pr^{1/3} \quad (19)$$

In the aforementioned simulation runs, RANS models are used for high Reynolds number simulations; the small eddies are represented by Reynolds stress terms based on closure models. To show the pore-scale mixing effect, a Large Eddy simulation (LES) is conducted. The Dynamic Smagorinsky-Lilly Model based on [18] is used for sub-grid scale modeling. $8,264,256$ grid cells are used with gradually shrinking cells towards the wall; the maximum $y +$ value of the cells adjacent to the wall is 2.59 . The time step size is 10^{-5} s . Though the average motion of the fluid is along the plates in the x direction, the pore-scale fluid exhibits strong unsteadiness and mixing effects, characterized by the cascade of eddies generated as a result of the fluid being periodically interrupted by the plates (Fig. 5(a)). The maximum local wall heat flux is found at the edges of the frontal areas due to impingement of fluid onto these surfaces (Fig. 5(b)). The flow is effectively cooled by the interrupted-plate medium.

3.2.3. Thermal dispersion analyses for a fixed shape

When volume averaging is applied to the energy equation, a thermal dispersion term appears. It is associated with spatial variations of velocity and temperature within the pores of the matrix. It is anisotropic in that a value found for streamwise dispersion is different than a value found for cross-stream dispersion. In this section, the streamwise thermal dispersion term for x -direction flow (see Fig. 3(c)) is calculated based on CFD runs on the REV

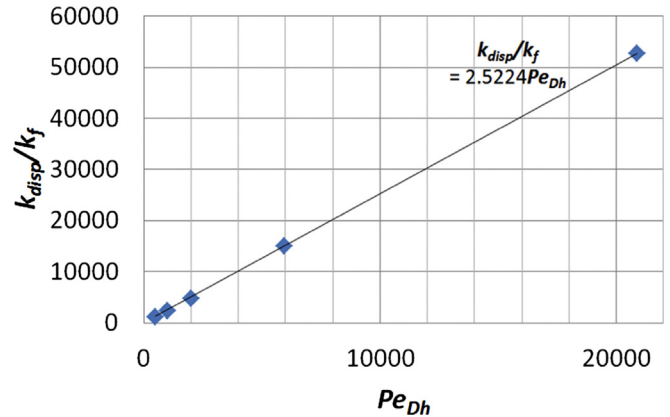


Fig. 6. Streamwise dispersion conductivity for x -direction flow.

model for the interrupted plate insert and $l = 7.5 \text{ mm}$, $t = 0.55 \text{ mm}$ and $2b = 2.75 \text{ mm}$. The term is:

$$-\rho c_p \langle \tilde{u} \tilde{T} \rangle^f = -\rho c_p \frac{1}{V_{REV, f}} \int_{V_{REV, f}} (\vec{u} - \langle \vec{u} \rangle^f) (T_f - \langle T_f \rangle^f) dV \quad (20)$$

The streamwise thermal dispersion term is modeled using the dispersion conductivity, k_{disp} ,

$$-\rho c_p \langle \tilde{u} \tilde{T} \rangle^f = k_{disp} \frac{\partial \langle T_f \rangle^f}{\partial x} \quad (21)$$

The dispersion conductivity is normalized based on the air conductivity and plotted against the Peclet number. The results are shown in Fig. 6. The Peclet number dependency is expected since dispersion transport is by mechanisms that are velocity dependent. One of the dispersion mechanisms is, by nature, similar to eddy transport in turbulent flow. For porous media, the “eddies” are within the pores and scale on pore size.

3.3. Experimental characterization of porous metal foams

The fine-scale geometry of the metal foam is randomly variable (over a feature size range) and essentially unknown to the numerical modeler. Computing the pressure drop, heat transfer and dispersion terms for representative, repeated geometries and

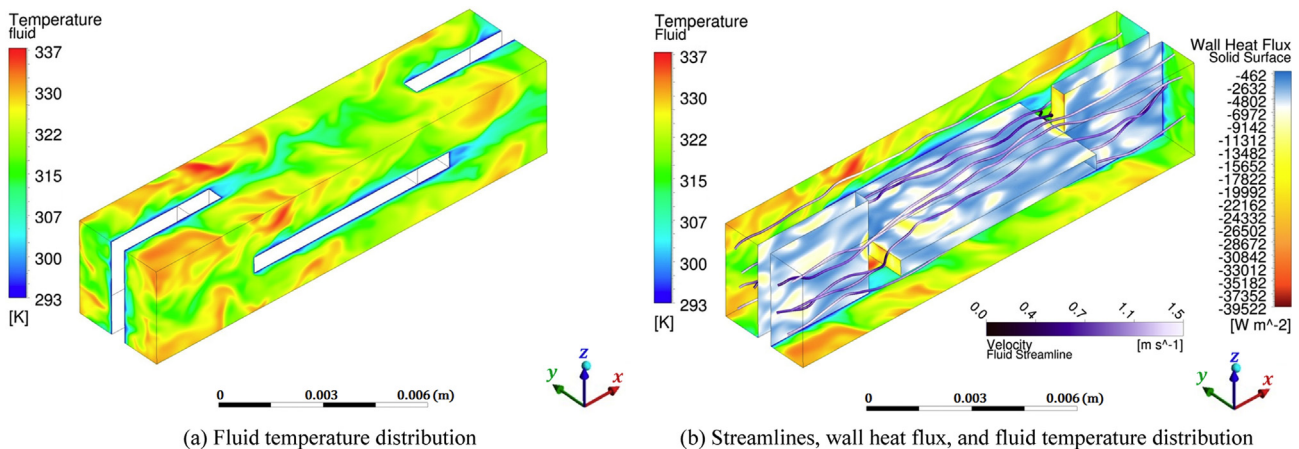


Fig. 5. Transient flow field of an LES simulation, $Re_{Dh} = 2872$ $l = 7.5 \text{ mm}$, $t = 0.55 \text{ mm}$, $2b = 2.75 \text{ mm}$, $\rho = 12.05 \text{ kg/m}^3$.

combining them statistically would be approximate and tedious. Thus, we have chosen to determine the pressure drop and interfacial heat transfer terms through experiments and simulation. The dispersion term for the foam matrix is discussed later.

Pressure resistance characteristics of the metal foams have been obtained by measuring pressure drop values across the metal foam section when air is blown through the foam at different velocities [19]. As shown in Fig. 7, the pressure drop shows a quadratic relation with Darcian velocity. Data from the measurements lead to the coefficients:

$$\begin{aligned} 10\text{PPI metal foam: } K &= 2.397 \times 10^{-7} \text{ m}^2, F = 0.140 \\ 40\text{PPI metal foam: } K &= 9.913 \times 10^{-8} \text{ m}^2, F = 0.181 \end{aligned}$$

The indicator PPI refers to the nominal number of pores per inch (per 25.4 mm), a frequently-used descriptor of these foams. They both have a porosity value of 93%.

A number of heat transfer correlations for porous media are available from the literature. To choose one that is most suitable, another experiment is done [19]. A schematic of the experimental setup is shown in Fig. 8. Water is circulated in the clockwise direction of the figure and pumped into the lower section of the liquid-piston compression chamber to compress air. Two experimental runs each are conducted with 10PPI and 40PPI metal foam inserts. Instantaneous values of volume and pressure are measured and instantaneous gas-volume-average temperatures are calculated using the ideal gas law. The experimental runs were simulated using the zero-dimensional model by substituting different heat transfer correlations found in the literature and the results were compared with the experimental results, as shown in Fig. 9. A modified version of the Kamiuto heat transfer correlation was selected, as discussed in Ref. [19].

$$\frac{h_v d_m^2}{k} = 0.996 \left(\frac{\rho u_{Da} d_m}{\mu} Pr \right)^{0.791} \quad (\varepsilon = 0.93) \quad (22)$$

where u_{Da} is the Darcian velocity, the velocity that the fluid would have in the absence of the solid.

3.4. Simulations of a compressor filled with metal foam

Numerical modeling of the metal-foam-inserted liquid-piston compressor combines Volume of Fluid (VOF) and two energy equations, one for the fluid and a second for the solid. Thus, we solve for energy transport in the fluid and solid phases and couple the two through the interfacial heat transfer correlation. The VOF method tracks the instantaneous location of the water/air interface in the compression chamber. Let subscripts 1 and 2 represent air and water, respectively. The continuity equations for air and water are:

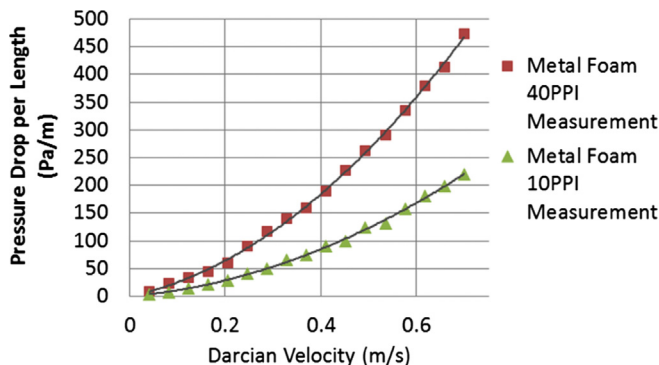


Fig. 7. Pressure drop measured from the experiments [19].

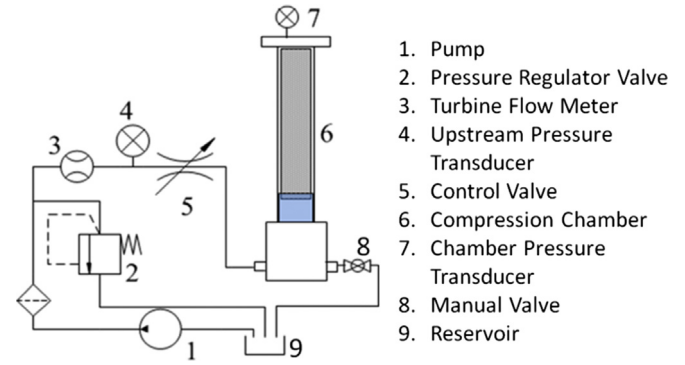


Fig. 8. Schematic of the experimental setup used to select the optimum open-cell metal foam heat transfer correlation [19].

$$\frac{\partial \alpha_1 \langle \rho_1 \rangle^f}{\partial \tau} + \nabla \cdot \left(\alpha_1 \langle \rho_1 \rangle^f \langle \vec{u} \rangle^f \right) = 0 \quad (23)$$

and

$$\frac{\partial \alpha_2}{\partial \tau} + \nabla \cdot \left(\alpha_2 \langle \vec{u} \rangle^f \right) = 0 \quad (24)$$

Momentum and energy equations are solved for the immiscible fluid mixture.

$$\frac{\partial \langle \rho \rangle^f \langle \vec{u} \rangle^f}{\partial \tau} + \nabla \cdot \left(\langle \rho \rangle^f \langle \vec{u} \rangle^f \langle \vec{u} \rangle^f \right) = -\nabla \langle p \rangle^f + \nabla \cdot \langle \vec{\tau} \rangle^f + \langle \rho \rangle^f \vec{g} + \vec{S}_m \quad (25)$$

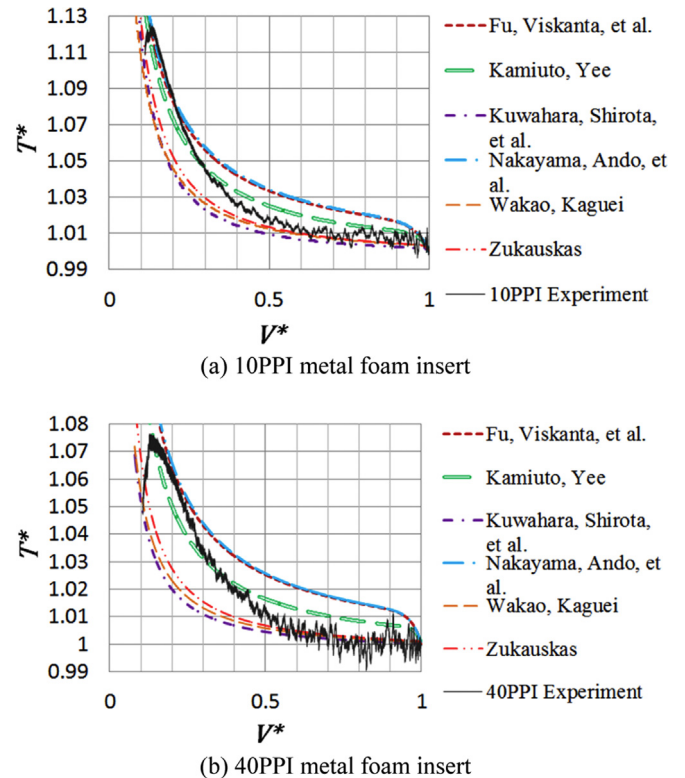


Fig. 9. Comparison of experimental results to the Zero-D model solutions obtained by using different heat transfer correlations [19]. (Variables are non-dimensionalized by initial values). Correlations: Fu et al. [20], Kamiuto and Yee [21], Kuwahara et al. [22], Nakayama et al. [23], Wakao and Kaguei [24], and Zukauskas [25].

$$\text{where } \langle \rho \rangle^f = \alpha_1 \langle \rho_1 \rangle^f + \alpha_2 \langle \rho_2 \rangle^f \quad (26)$$

The stress tensor, $\bar{\tau}$, is based on viscosity of the mixture of air and water in nodes that contain the liquid–gas interface.

$$\mu = \alpha_1 \mu_1 + \alpha_2 \mu_2 \quad (27)$$

The momentum sink term, $S_{m,x} = \partial p / \partial x$, is given by Eq. (12). The energy equation for the fluid is.

$$\begin{aligned} & \frac{\partial \varepsilon (c_p \langle \rho \rangle^f \langle T \rangle^f)}{\partial t} + \varepsilon \nabla \cdot (\langle \bar{u} \rangle^f \langle \rho \rangle^f c_p \langle T \rangle^f) \\ & = \varepsilon \nabla \cdot k_f \nabla \langle T \rangle^f - \nabla \cdot \rho c_p \langle \bar{u} \rangle^f \tilde{T} + h_v (\langle T_s \rangle^s - \langle T \rangle^f) + \varepsilon \frac{\partial \langle p \rangle^f}{\partial t} \end{aligned} \quad (28)$$

where

$$\langle \rho \rangle^f c_p = \alpha_1 \langle \rho_1 \rangle^f c_{p,1} + \alpha_2 \langle \rho_2 \rangle^f c_{p,2} \quad (29)$$

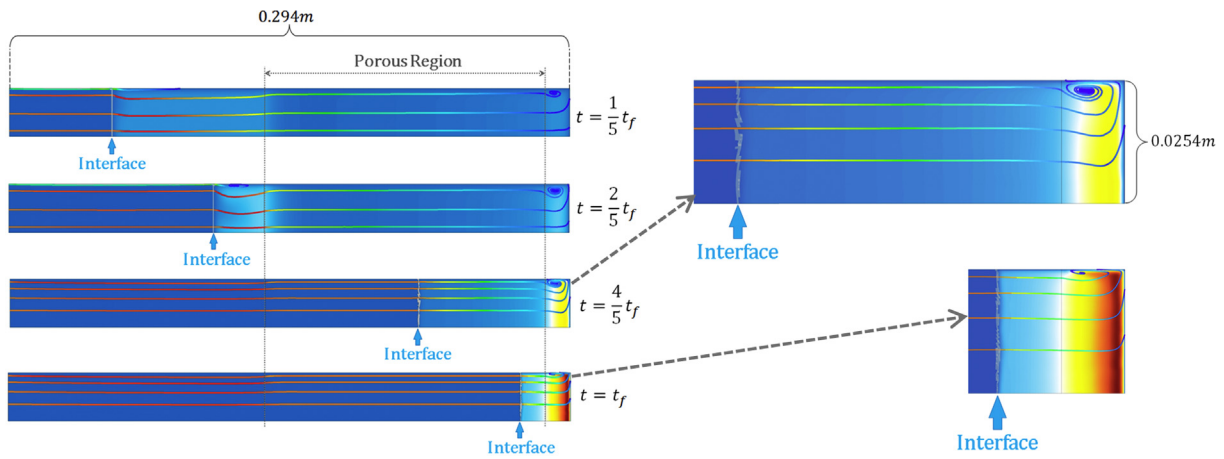
and

$$k_f = \alpha_1 k_1 + \alpha_2 k_2 \quad (30)$$

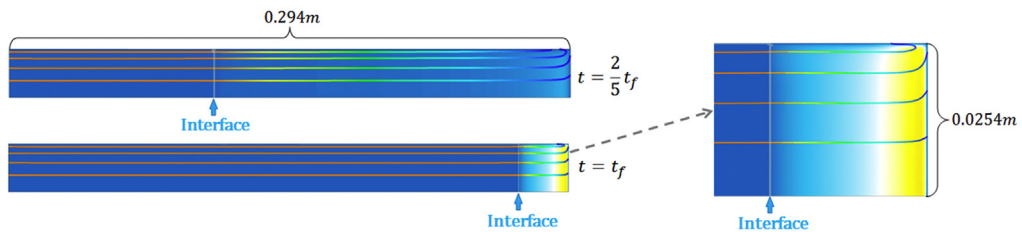
The energy equation for the solid is.

$$(1 - \varepsilon) \frac{\partial}{\partial t} (\langle \rho_s \rangle^s c_s \langle T_s \rangle^s) = (1 - \varepsilon) \nabla \cdot k_s \nabla \langle T_s \rangle^s - h_v (\langle T_s \rangle^s - \langle T \rangle^f) \quad (31)$$

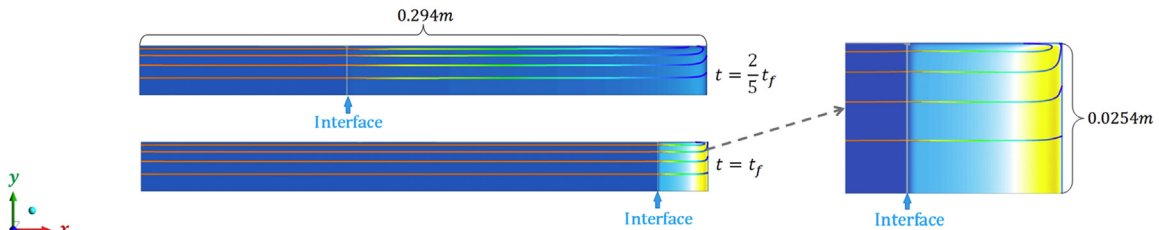
In the study of ref. [19], thermal dispersion term was neglected ($\nabla \cdot \rho c_p \langle \bar{u} \rangle^f \tilde{T} = 0$). It shows that when only a portion of the chamber



(a) Metal foam partially inserted in the chamber, no dispersion [19]



(b) Metal foam fully-inserted in the chamber, no dispersion [19]



(c) Metal foam fully-inserted in the chamber, with dispersion term included

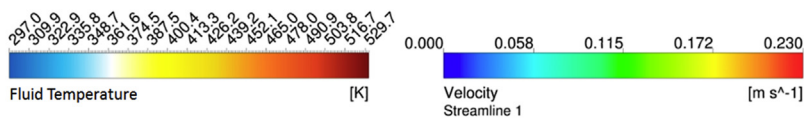


Fig. 10. CFD simulations on the chamber with metal foam insert (note that up is to the right in this figure). 10PPI metal foam. $U_0 = 0.206$ m/s, $t_f = 1.3$ s. The chamber length and radius are respectively: 0.294 m and 0.0254 m. (Shown on the left is the entire chamber region; shown on the right is a scaled-up view of the air region).

is occupied by porous insert, flow is stable in the porous insert region but vortices develop elsewhere (Fig. 10(a)); when the entire length of the chamber is occupied by insert, velocity streamlines are smooth (see Fig. 10(b)). This indicates that optimization of the distribution of porous inserts in the chamber may be done to achieve minimal temperature rise of air during compression, and best compression efficiency.

In the present study, a CFD simulation is conducted with the thermal dispersion term included. The chamber is fully occupied by 10PPI metal foam. The thermal dispersion term is modeled based on dispersion conductivity.

$$-\rho c_p \langle \tilde{u} \tilde{T} \rangle^f = \bar{k}_{dis} \nabla \langle T \rangle \quad (32)$$

Kuwahara and Nakayama [26] proposed a closure model for thermal dispersion for 2-D cross flow over small square rods:

$$\frac{k_{dis,XX}}{k_f} = 0.022 \frac{Pe_D^2}{(1-\epsilon)} \quad (Pe_D < 10) \quad (33)$$

$$\frac{k_{dis,YY}}{k_f} = 0.022 \frac{Pe_D^{1.7}}{(1-\epsilon)^{1.4}} \quad (Pe_D < 10) \quad (34)$$

$$\frac{k_{dis,XX}}{k_f} = 2.7 \frac{Pe_D}{\epsilon^{1/2}} \quad (Pe_D > 10) \quad (35)$$

$$\frac{k_{dis,YY}}{k_f} = 0.052(1-\epsilon)^{1/2} Pe_D \quad (Pe_D > 10) \quad (36)$$

where XX is the local streamwise direction and YY is the local cross-stream direction. This model is implemented in the CFD simulation in the present study. The result shows that adding thermal dispersion to the energy equation does not affect the flow field much, as shown in Fig. 10(c). Close scrutiny shows that at the end of compression, the case with dispersion included has a smaller maximum local temperature than the case neglecting dispersion.

Transient values of bulk temperature of air are compared between simulations with and without the dispersion term (Fig. 11). Results of both cases show that, though the compression time is short and isentropic compression would lead to a temperature of 575 K, the insert is effective in suppressing temperature rise. Further, they show that during the latter period of compression, calculated temperatures of the case with dispersion are less, because of larger thermal gradients and higher heat transfer.

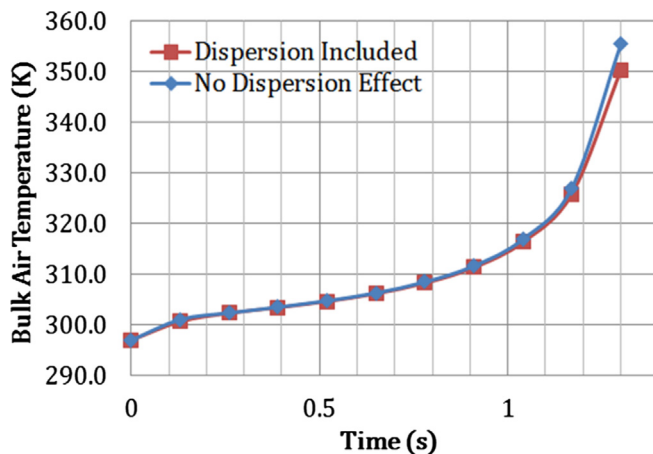


Fig. 11. Bulk temperature of air.

Thermal dispersion has a greater influence on the maximum bulk temperature when temperatures and temperature gradients rise near the end of compression, particularly for large pressure ratios. When thermal dispersion effect is included, the computed bulk air temperature at the final compression state is 5 K less than when the dispersion effect is not included. The local maximum temperature values at the final compression state for the case without dispersion effect and the one with dispersion effect are respectively: 417.2 K and 428.3 K. This temperature difference caused by dispersion has a small effect on the compressor performance for the cases investigated in the present study, since this temperature difference is small compared to the total amount of temperature rise during compression.

4. Conclusions

A compressed air energy storage system designed for use in wind turbine plants was introduced and the importance of thermal control during compression and expansion was emphasized. A compressor design was proposed and models for its analysis were presented. Noted in modeling was a need for sub-models to characterize pressure drop in the porous insert, heat transfer from fluid to solid phases of the insert and thermal dispersion within the insert. Such models were presented for two candidate inserts, one based on interrupted plates and another based upon metal foams. The models came from detailed computation of a representative elementary volume, in the case of the interrupted plate, and from experiments, in the case of the foam. Computational results using these models for the foam insert case were presented. The inserts are very effective in suppressing temperature rise. The metal foam kept the bulk temperature below 360 K if it occupied the full chamber length; and, for reference, the adiabatic bulk temperature would have been 575 K if no heat transfer medium had been introduced. Computed results that compare a case with the insert occupying the full chamber length and a case having the insert occupying only a portion of the chamber length show that the insert stabilizes the flow, where present; but beyond the insert there is a tendency for development of secondary flows. The diameter of the secondary flow vortex scales on the thickness of the non-porous region. A study in which dispersion was added to the analysis showed that effect of dispersion is small and in the direction of reducing the maximum average temperature reached during compression. Dispersion weakens temperature gradients in the fluid, increasing near-wall gradients and heat transfer to the walls. For the pressure ratio studied, dispersion has reduced the maximum local temperature by 10 K.

Acknowledgements

This work is supported by the National Science Foundation under grant NSF-EFRI #1038294 and University of Minnesota, Institute for Renewable Energy and Environment (IREE) under grant: RS-0027-11. The authors thank also the Minnesota Supercomputing Institute for the computational resources used in this work.

References

- [1] P. Sullivan, W. Short, N. Blari, Modeling the benefits of storage technologies to wind power, in: American Wind Energy Association (AWEA) WindPower 2008 Conference, Houston, Texas, June, 2008.
- [2] J. Garrison, C.W. King, M.E. Webber, R. Baldick, Twenty percent wind, energy storage, and Texas grid, in: Proceedings of the ASME 2013 International Mechanical Engineering Congress and Exposition, San Diego, Nov. 2013.
- [3] US Offshore Wind Collaborative, U. S. Offshore Wind Energy: a Path Forward, A Working Paper of the U. S. Offshore Wind Collaborative, October, 2009.

- [4] C. Bullough, C. Gatzen, C. Jakiel, M. Koller, A. Nowi, S. Zunft, Advanced adiabatic compressed air energy storage for the integration of wind energy, in: *Proceedings of the European Wind Energy Conference, EWEC 2004*, Nov. 22–25, London, UK, 2004.
- [5] M. Nakhamkin, M. Chiruvolu, M. Patel, S. Byrd, R. Schainker, Second generation of CAES technology – performance, operations, economics, renewable load management, green energy, in: *POWER_GEN International Conference*, Las Vegas, NV, Dec. 2009.
- [6] C. Langness, D. Kolsky, T. Busch, C. Davidson, C. Depcik, Proof-of-concept combined shrouded wind turbine and compressed air energy storage system, in: *Proceedings of the ASME 2013 International Mechanical Engineering Congress and Exposition*, San Diego, Nov. 2013.
- [7] P. Y. Li, J. Van de Ven, C. Shancken, Open accumulator concept for compact fluid power energy storage, in: *Proceedings of the ASME 2007 International Mechanical Engineering Congress and Exposition, IMECE*, 2007.
- [8] P.Y. Li, E. Loth, T.W. Simon, J.D. Van de Ven, S.E. Crane, Compressed air energy storage for offshore wind turbines, in: *2011 International Fluid Power Exhibition (IFPE)*, Las Vegas, NV, March, 2011.
- [9] J. Van de Ven, P.Y. Li, Liquid piston gas compression, *Appl. Energy* 86 (10) (2009) 2183–2191.
- [10] R. Tew, M. Ibrahim, D. Danila, T.W. Simon, S.C. Mantell, L. Sum, D. Gedeon, K. Kelly, J. Mclean, G. Wood, S. Qiu, A Microfabricated Involute-foil Regenerator for Stirling Engines, *NASA/CR-2009-215516*, 2009.
- [11] X. Qi, Q. Guo, Analysis of start-up characteristics of a honeycomb regenerator, *Res. J. Appl. Sci. Eng. Technol.* 6 (20) (2013) 3784–3797.
- [12] G. Kolios, D. Luss, R. Garg, G. Viswanathan, Efficient computation of periodic state of cyclic fixed-bed processes, *Chem. Eng. Sci.* 101 (2013) 90–98.
- [13] C. Aprea, A. Greco, A. Maiorino, A dimensionless numerical analysis for the optimization of an active magnetic regenerative refrigerant cycle, *Int. J. Energy Res.* 37 (2013) 1475–1487.
- [14] C.J. Sancken, P.Y. Li, Optimal efficiency-power relationship for an air moto-compressor in an energy storage and regeneration system, in: *Proceedings of the ASME 2009 Dynamic Systems and Control Conference, DSCC*, 2009.
- [15] C. Zhang, T.W. Simon, P.Y. Li, Storage power and efficiency analysis based on CFD for air compressors used for compressed air energy storage, in: *Proceedings of the ASME 2012 International Mechanical Engineering Congress and Exposition*, Houston, TX, Nov. 2012.
- [16] C. Zhang, F.A. Shirazi, B. Yan, T.W. Simon, P.Y. Li, J. Van de Ven, Design of an interrupted-plate heat exchanger used in a liquid-piston compression chamber for compressed air Energy storage, in: *Proceedings of ASME 2013 Summer Heat Transfer Conference*, Minneapolis, MN, July 2013.
- [17] F.R. Menter, R.B. Langtry, S.R. Likki, Y.B. Suzen, P.G. Huang, S. Volker, A Correlation Based Transition Model Using Local Variables Part 1 – Model FORMULATION,” *ASME-GT2004-53452*, 2004.
- [18] D.K. Lilly, A proposed modification of the Germano subgrid-scale closure model, *Phys. Fluids* 4 (1992) 633–635.
- [19] C. Zhang, J.H. Wieberdink, F.A. Shirazi, B. Yan, T.W. Simon, P.Y. Li, Numerical investigation of metal-foam filled liquid piston compressor using a two-energy equation formulation based on experimentally validated models, in: *Proceedings of the ASME 2013 International Mechanical Engineering Congress and Exposition*, San Diego, CA, Nov. 2013.
- [20] X. Fu, R. Viskanta, J.P. Gore, Measurement and correlation of volumetric heat transfer coefficients of cellular ceramics, *Exp. Therm. Fluid Sci.* 17 (1998) 285–293.
- [21] K. Kamiuto, S.S. Yee, Heat transfer correlations for open-cellular porous materials, *Int. Comm. Heat Mass Transf.* 32 (2005) 947–953.
- [22] F. Kuwahara, M. Shirota, A. Nakayama, A numerical study of interfacial convective heat transfer coefficient in two-energy equation model for convection in porous media, *Int. J. Heat Mass Transf.* 44 (2001) 1153–1159.
- [23] A. Nakayama, K. Ando, C. Yang, Y. Sano, F. Kuwahara, J. Liu, A study on interstitial heat transfer in consolidated and unconsolidated porous Media, *Heat Mass Transf.* 45 (11) (2009) 1365–1372.
- [24] N. Wakao, S. Kaguei, *Heat and Mass Transfer in Packed Beds*, Gordon and Breach, New York, 1982, pp. 243–295.
- [25] A.A. Zukauskas, Convective heat transfer in cross-flow, in: S. Kakac, R.K. Shah, W. Aung (Eds.), *Handbook of Single-phase Convective Heat Transfer*, Wiley, New York, 1987.
- [26] X.F. Kuwahara, A. Nakayama, Numerical determination of thermal dispersion coefficients using a periodic porous structure, *Technical Notes, J. Heat Transf.* 121 (Feb. 1999) 160–163.